Go Back

Measured System Performance
and
Diagnostic Testing
Report

U.S. Air Force Academy, Colorado Commissary

Prepared for:

Defense Commissary Agency (DeCA)
Directorate of Facilities
Facilities and Programs Division
Fort Lee, VA

Prepared by:

U.S. Army Construction Engineering Research Laboratories
Champaign, IL

Table of Contents

Nomenclature

ACC: air-cooled condenser
AFS: air flow switch (proof)
AH-1: air handler #1 (bakery zone)
AH-2: air handler #2 (administrative zone)
ASL: above sea-level
CHWS/R: chilled water supply/return
Contractor: DeCA contract maintenance personnel
CT: current transducer
DB: dry-bulb
DD-1: desiccant dehumidification system (main store air handler)
DeCA: Defense Commissary Agency
DHW: domestic hot water
ECO: energy conservation opportunity
EPRI: Electric Power Research Institute
EUI: energy use intensity (usually in kWh/sf or kBtu/sf)
HWS/R: hot water supply/return
LT: low- temperature (22°F or lower evaporator temp /32°F or lower case temp)
MDL: MicroDataLogger ®
MT: medium temperature (40°F or lower evaporator temp /50°F or lower case temp)
NBECS: National Building Energy Consumption Survey
OA: outside air
Period: the period of measurement testing, in this case 7 June to 28 June 1996
PVT: performance verification testing
RH: relative humidity
RMCS: refrigerant management and control system (digital)
ReCX: recommissioning
TAB: test and balance
WB: wet-bulb

Summary of Recommended Action Items

1. Negotiate lower flat-rate electricity charge with Academy Energy Office.
2. Reduce DD-1 supply air CFM.
3. Implement DD-1 night setback.
4. Restore heat reclaim function to DD-1 coils.
5. Repair DD-1 fire damper actuator and linkages and PVT fire suppression system.
6. Increase deadband on heating and cooling of DD-1 zone.
7. Repair desiccant unit:
    a. Repair tubing from flow proof switch 1AFS.
    b. Correct fan sequence logic on desiccant unit (if needed).
9. Recalibrate humidistat and lower RH setpoint to 35%.
10. Implement night/off-hour setback on admin zone AH-2.
11. Implement economizer control on AH-2.
12. Consider installing dedicated rooftop DX unit for AH-2.
13. Get chilled water under on/off control by implementing OA lockout. This strategy will substantially reduce compressor cycling. Also, night setback of AH-1 and AH-2 will contribute to reducing cycling.
14. Investigate additional strategies for optimizing oversized chiller operation.
15. Address oversizing of packaged chiller. Consult factory rep for potential short-term operational scenarios prior to startup next season.
16. Downsize chilled water pumps (or trim impeller). Dependent on package chiller oversizing decision.
17. Get boilers under on/off control by implementing OA lockout.
18. Implement lead/lag strategy for boilers.
19. Implement hot water reset strategy to save heating energy during spring and summer operation.
20. Implement boiler staging (upon boiler replacement).
21. Trim impellers on HWS to match maximum load requirement.
22. Repair condensate leak at boiler B-2 exhaust.
23. Disconnect antisweat heaters (contractor had conflicting information here).
24. Shut off case lights during unoccupied hours (reactivate existing controls).
25. Set Danfoss control system to monitor compressor daily cycles.
26. Exhaust fan (EF-8) in mezzanine not operating. Relay disabled. Investigate reason and reconnect.
27. Investigate circuits 10 and 14 for problems relating to excessive sources of moisture at or near the load.
28. Reduce light levels during off hours.

1.0 Introduction: Commissary and Supermarket Recommissioning (ReCX)

Purpose of Study

The primary purpose of this study is to examine opportunities for cost savings at the USAF Academy Commissary through the identification of problems with the design, construction, operation, and maintenance of the facility as it currently exists. These problems, known in the building commissioning trade as *deficiencies*, are unfortunately common to almost all buildings. Their existence is due to many factors, including:

- communications breakdowns throughout the design and construction process;
- short-sighted cost savings and value engineering measures;
- inadequate funding for the design, construction, and acceptance measures required to properly construct a building;
- insufficient design review, construction inspection, and verification of performance of the building and its interrelated systems

The identification of deficiencies after a building is built and has been in operation, along with documentation and implementation of improvements to address those deficiencies, is often called *recommissioning (ReCX).* A related and more often-used term - commissioning - is <u>traditionally</u> considered to be simply the task of calibrating and performance verification testing (PVT) individual pieces of HVAC and refrigeration equipment. This term is finally being revised and accepted by the building industry to include whole-building system operational testing, and owner representative engineering services from project concept design through beneficial occupancy, sometimes even well into the operation and maintenance cycle. The goal of building commissioning and recommissioning is the same: to provide a building owner and operating staff an efficient, cost-effective, healthy, and productive environment, and to provide much-needed feedback to designers as to what aspects of their designs do and do not work in the field.

Goals for Defense Commissary Agency Recommissioning

The goals for doing a commissioning-based study of commissaries are as follows:

- to provide a low cost, proactive approach for simultaneously addressing energy and operational problems;
- to assist in bringing a facility into its optimal state of performance prior to implementation of more costly energy conservation measures (ECOs);
- to identify chronic problems which may not be readily identifiable by store personnel or contractors. Acute or intermittent problems - such as equipment failure which include alarm-setting events - may also be identified but are typically fixed by

maintenance contractors. The intent of this effort is not to assign blame, but to identify problems that might otherwise go unnoticed and to properly address them.

- to gather information to provide quantitative feedback to DeCA design and contract maintenance personnel;
- to contribute to the development of a commissary/supermarket ECO computer model;
- to determine a consistent baseline for the installation of energy efficiency measures;
- to begin development of future performance goals for DeCA facilities.

On Recommissioning, the Energy Audit, and ECO Analysis

In the 1970s - and again in the 1990s - the facility energy audit has been a useful tool in determining ways to identify ECOs for existing buildings. In government operations, funding is often made available by various agencies to purchase and install new energy-efficient technologies. Numerous programs have spent millions of dollars to install these technologies, sometimes in buildings that were constructed recently enough to have included them in the first place. Upgrade of buildings with more efficient equipment is an important means to improving the cost-effectiveness of building operation. However, recent studies of ECO project performance indicate that many are not performing as well as anticipated. The dominant reason for this is that the projects are not themselves commissioned properly upon installation, and - more importantly - they are often installed on buildings that are erroneously assumed to be operating properly.

As government-funded energy efficiency programs decline, more pressure is put on agency energy and operations managers to do more with less. Building recommissioning is an appropriate first step toward efficient building operation, often requiring little or no capital equipment expenditures to achieve cost savings and efficient operations. The benefits of recommissioning include longer equipment life, more efficient operation of existing equipment, a better trained operations and maintenance staff, and improved occupant comfort and productivity. And, after a building is recommissioned - especially if an instrumented approach is used - there is also a well defined baseline in which to calculate the efficacy of future ECO projects and occupant education programs.

Types of Deficiencies

Deficiencies identified through the recommissioning processed can be classified in the following categories or with the following origins:

A) Design deficiencies
B) Construction deficiencies
C) Operation deficiencies
D) Maintenance deficiencies

Design deficiencies are those that arise during the design phase, often from poor communication between the project architects and various members of the engineering staff, or from inadvertent or incomplete specification of inappropriate technologies and control sequences. Commonly identified examples include equipment oversizing, conflicting design specifications, and incomplete control sequences.

The construction process, with its many contractors and subcontractors each having to work and schedule their tasks around each other, is a common source of deficiencies. Examples of construction deficiencies include unattended punchlist items, inappropriate equipment substitution (often due to contractor cost savings), inadequate quality assurance checks, and inefficient whole-building system control interactions.

Operational deficiencies are those that occur or develop due to improper operation of otherwise well-designed and constructed building systems. These are most commonly system control related issues (although ineffective controls can also be traced to design or construction origins as well). Often, building staff are not taught to properly set thermostats, or maintenance personnel remove a piece of equipment from automatic operation in order to temporarily override some aspect of its operation, never to return it to its original mode.

Maintenance deficiencies are similar to operational, and the exact origin of a deficiency can be murky. For the purposes of this study, a maintenance deficiency is one that has its origin in equipment malfunction or breakdown. The most common maintenance deficiencies are repetitive equipment malfunctions that go unsolved or unaddressed. Sources of these problems can primarily be traced to inadequate diagnostic equipment and time to properly diagnose and correct a problem. Also, maintenance personnel often lack adequate O&M documentation and do not have the opportunities for training. Poor O&M documentation often consists merely of catalog cuts and pieces of original design documents.

Experience shows that two of the most beneficial results of the recommissioning process should be (i) the development of a useful O&M document that reflects the actual installed equipment, a *working* sequence of operation, vendor phone numbers for technical support, and addresses the site specific problems in a building as seen by the O&M staff; and (ii) on-site training of maintenance personnel using the actual equipment in the building.

Acknowledgments

The on-site staff, particularly Chris Burns and Larry Price, were very supportive and provided substantive input into the still evolving commissary recommissioning process. Information needed by the USACERL/EMC audit team was readily available. Complete access was available at all times and was much appreciated. The Academy commissary provided a unique site in which to test out various metering strategies and learn about the unique nature of supermarket refrigeration and dehumidification systems.

USACERL acknowledges the work of Architectural Energy Corporation, the Electric Power Research Institute (EPRI) and Mr. Mark Arney, for his diligence and expertise in use of the MDL datalogger and the HDCS (Beta) software.

USACERL would also like to acknowledge the efforts and resources of EMC Engineers, Denver, CO. EMC helped to obtain a great deal of site information and had full sized drawings reproduced at their own expense to help in the development of the metering plan. They also provided engineering support through verbal communications and discussion of various commissary-related issues.

2.0 Overview of Energy Systems and Operations at the USAF Academy Commissary

Building Design and Space Allocation

The USAF Academy commissary was designed in FY 90, built in FY92 and opened for business in August 1992. Cromwell Architects/Engineers (Little Rock, AR) was the design agent. The approximate building square footage allocated by function in provided in Table 2-1.

Function	**Square Footage**
Sales and Checkout	30,600
Warehouse/Semi-Perishable Receiving (H,V only)	10,200
Conditioned - Meat (prep, staging, storage)	4,300
Administrative (all offices; AHU-2 serves 3000 sf)	8,300
Mezzanine/Mechanical (H,V only)	3,000
Conditioned - Frozen Food (storage)	2,200
Conditioned - Produce (prep, storage)	1,700
Conditioned - Dairy (storage)	1,700
Bakery	1,300
Total:	**63,300**

Table 2-1. USAF Academy Commissary square footage breakout.

Climate Overview

The weather at the USAF Academy presents a challenge for the both the mechanical designer and the personnel responsible for operating and maintaining the refrigeration and HVAC systems. It high-altitude location (about 7,200 ft. above sea-level) and the highly variable nature of weather along the eastern slope of the Rocky Mountains can provide occasional weekly temperature differences in excess of 80°F (one worker at the store reported daytime high temperatures in the 70's followed by cold weather reaching -10°F, and a 48 hour stretch which did not exceed 0°F, all within a 9-day period in early January 1997). Design weather data for the Academy include the following statistics:

Elevation: 7,200' ASL (approx. at site)
Mean daily temperature range: $\Delta T = 30°F$
Median of annual extremes: 96°F (high); -10°F (low)

Design DB temperatures:

Cooling:

1%: 91°F (high) / 58°F (low)

2.5%: 88°F / 57°F

5%: 85°F / 57°F

Heating:

97.5%: 2°F

99%: -3°F

Design WB temperatures:

1%: 63°F

2.5%: 62°F

5%: 61°F

Refrigeration System Overview

The total installed case load (design) is 89.3 tons (1,071,400 Btu/h). The refrigeration system consists of 4 multiplexed unequal/parallel racks of reciprocating Carlyle compressors built-up by the Tyler Corp. (Tyler *Equalizer* Series). Racks 1 & 2 serve low temperature refrigeration circuits, and racks 3 & 4 serve medium temperature loads. The system is operated using a fixed-head pressure strategy, with one multi-fan air-cooled condenser unit (ACC) serving each rack of compressors. Fans on the ACC are modulated with outside air to achieve fixed head pressure operation. A heat reclaim system is available prior to condensing to extract waste heat from the hot refrigerant gases delivered by the compressor racks. This waste heat can be sent to two heat reclaim storage tanks for preheating domestic hot water (DHW), or to coils on the main air handling system which use the hot gas to provide auxiliary heating of the supply air stream or to regenerate a desiccant dehumidification system (see 'HVAC system' below). Specifically, heat reclaim from Rack 1 is designed to provide DHW preheating or regenerate the desiccant wheel when active; Rack 2 provides DHW preheating or auxiliary heating of return air; and Racks 3 & 4 both are designed for heat reclaim to provide only for auxiliary heating of return air. A Danfoss refrigeration management and control system (RMCS) controls operation of both the refrigeration system and the associated HVAC system. During the course of the study (July 1996) the refrigerant used in the low temp racks was changed from R-502 to R-404a. A summary with detailed refrigeration system and circuit information is provided in Appendix A.

HVAC System Overview

A 65-ton packaged chiller (Dunham-Bush) and 2 - 880,000 Btu/hr output (704,000 @ 7,200 ft. elev.) packaged low-pressure hot water boilers (Lochinvar Corp.) provide chilled and hot water respectively to coils associated with 3 separate air handling units (AHU) and 6 fan coil (FC) units which provide both heating and cooling to various offices and breakrooms throughout the store. The main air handling unit (DD-1) serves the retail floor area, and has a built-in desiccant system (Munters) to dehumidify a blend of outside and store return air when necessary. Another air handling unit (AHU-2) serves the administrative office space. A third unit (AHU-1) serves the smaller bakery/deli area. There are 21 hot water-fired unit heaters (UH) located in the warehouse and receiving areas and other areas where heating only is required. A stand-alone 3-ton DX air conditioning system provides year-around cooling to the ADP/computer room. Ventilation requirements are met with outside air intakes on DD-1 and AHUs 1 & 2, in addition to 12 dedicated exhaust fans (EF) ranging in design capacity from 60 to over 15,000 CFM.

Lighting System Overview

The lighting system is primarily 277V, T-12 fluorescent. Both 8' shop-type luminaires and 2'x4' troffer lighting is used for retail zone, warehouse, and administrative office lighting.. Aside from lighting controls, which are often used in supermarkets and are controlled by the RMCS system, there are often few lighting items to be measured and analyzed . Of note in this building is a substantial amount of daylight present through the use of vertical clerestory glazings (transparent to the north, translucent to the south). This situation could provide an ideal opportunity to use energy-saving daylight dimming controls to control store lighting.

One existing deficiency of note was pointed out by DeCA HQ and the store staff. A long-standing problem exists with the building control system that prohibits the store lighting to be turned down to 50% automatically during unoccupied hours.

EMC Engineers, Inc., is performing an extensive analysis of the store lighting systems as part of their energy audit of the Academy Commissary. Further analysis of lighting systems as part of a recommissioning effort is not warranted in this case.

Operations Overview

There was a change of contract maintenance provider on 1 June 96. Prior to this date, the contractor was Hussmann Corp. (Mr. Gary Baldwin); the new contractor is Scott-Polar (service provided by various personnel). There were some changes observed during the period from June to early September, but no obvious problems with the new contractor are indicated. In May 1996, the store appeared to be running extremely well with no obvious problems and interviews indicated a satisfied management staff. However, discussions with a number of store personnel in early September 1996 indicate rather extreme dissatisfaction with the new arrangement. Store supervisors felt that Scott-Polar is being called in more often than was Hussmann, and that there have been a lot of problems with defrost, particularly in the frozen food walk-in box. A walk-through of the area during the early September visit found 2 unit coolers completely frosted closed (fans was frozen and unable to turn); there was ice on the floor and an apparent attempt by store personnel to put cardboard on the floor to help with traction; and a large amount of hoarfrost had collected on surfaces near the vinyl curtains. According to store personnel, Scott-Polar has attempted to fix the problem a number of times, and is looking into insulating the ceiling above the cooler, and has also attributed the problem to high humidity in the store. Data taken during the period show no increase in humidity over previous years. Maintenance logs indicate some problems and repairs to drains in the frozen food walk-in.

Existing Conditions Overview

The walk-through and metering visits indicated an attractive, extremely well-run commissary. The equipment room was quite clean and no unsafe conditions were observed. Efficient energy-related practices by store staff were apparent, including use and understanding of load-line limits in cases, turning unused lights off, and closing of walk-in cooler doors. A few minor items were worn, such as door seals and plastic curtains. Commissary management was notified and scheduled replacement. Replacements were noted during a follow-up visit 30 days later. Efficient cold stock unloading, storage, and stocking procedures were noted, with some workers closing freezer doors behind them as they entered the walk-ins.

Display cases were clean and apparently well maintained. Monthly preventive maintenance is carried out by the contractors to inspect case lighting and fans, and to clean case coils when needed.

Refrigeration Management and Control System (RMCS) Overview

A Danfoss-EMC, Inc. ZX refrigeration management and control system (RMCS) provides digital control of both the refrigeration system and various HVAC functions. The points currently monitored by the RMCS are not entirely adequate for the level of detail needed for metered recommissioning. The RMCS is set up to track primarily refrigeration system suction, discharge, and condensing (head) pressures, as well as providing detailed measurement and control of cases and walk-in temperatures, and out-of-range/alarm conditions. Currently, use of the RMCS by store personnel involves printing a hardcopy report 4 times daily, and use of the computer terminal to track alarm conditions when they occur. The daily reports are kept on file in the store managers office. The local maintenance contractor also has access to the system from a remote site. Store officials have the capability to reset alarms, while the local maintenance contractor has the added ability to reset or add defrost schedules. Change of case setpoints can only be done by DeCA HQ via remote access at this time. A printout of the RMCS points list is provided in Appendix A. Discussion of specific setpoints and operation is provided in the detailed system discussions in Section 4.0 below.

Measurements and Metering Plan Overview

A system metering and component testing plan was developed to provide information on the major energy consuming systems and components. Time series data was collected using 28 small, 4-channel data acquisition systems [Model 201 MicroDataLogger (MDL), Architectural Energy Corporation, Boulder CO.]. The loggers were deployed throughout the facility for a 21-day period beginning at midnight 7-June 1996. The sample rate was a single unaveraged value taken at 2-minute intervals throughout the period for all measurements. The Danfoss RMCS system was manually polled on occasion to obtain case temperature data and head and suction pressure values, but could not be programmed to provide time series trend data. Individual equipment was also used for various tasks, including a Neotronics combustion efficiency analyzer, a Controlotron System 990 multipulse ultrasonic flow meter (clamp-on), a Fluke 41B power harmonics meter and scope, and a Solomat 510E HVAC diagnostics kit (temperature, relative humidity, hot wire anemometer, pitot tube, CO_2, CO, and tachometer). A overview of the measurement and metering plan used is provided below.

<u>Metered Recommissioning (3-week; 2-minute time-series data):</u>

1. Packaged chiller (CH-1)
2. Chilled water pumps and distribution
3. Packaged hot water boilers (B1 & B2)
4. Hot water pumps and distribution
5. Main air handling/desiccant unit (DD-1)
6. Administrative area air handling unit (AHU-2)
7. Refrigeration compressor Racks 1 through 4 (R1-4)
8. Electric defrost (Racks 1, 2, and 3)
9. Store ambient conditions - dry grocery/checkout
10. Store ambient conditions - cold aisles
11. Rooftop weather station

<u>Diagnostics and Functional Testing (single measurement):</u>

1. Chilled water supply flow rate (ultrasonic)
2. Hot water supply flow rate (ultrasonic)
3. Boiler combustion efficiency
4. Supply, return, and outside air flow rates
5. Real power measurements (corresponding to all current measurements)
6. Voltage and power factor measurements (corresponding to all current measurements)
7. Chilled water valve/actuator (PVT)
8. Hot water valve/actuator (PVT) (the frozen pea test)
9. Heat reclaim valve check at DD-1 (PVT)
10. HVAC logic check (as programmed in Danfoss RMCS)

Energy Disaggregation by End-Use:

1. Whole-building (120V/5A meter at main panel)
2. Refrigeration system (w/defrost breakout)
3. Lighting system (277/480V panels)
4. HVAC subsystem
5. Battery charging stations
6. Misc. Equipment (all 120/208V loads, inc. office equipment)

Systems not analyzed for this report:

1. Domestic water heating (DHW) system
2. ACC 1-4 (serving refrigeration racks 1 - 4)
3. ACC-5 (serving packaged chiller)
4. AHU-1 - Bakery
5. AHU-3 - ADP Room
6. Exhaust systems
7. Fan coil operation (distributed small offices)
8. Warehouse heating and ventilation
9. Compressed air (for pneumatic actuation)

The RMCS can 'trend' user selected points to provide historical data of system performance. This capability is quite useful and can supplant the need for some of the datalogging in a metered recommissioning effort. However, the points measured by most RMCS systems often are not adequate when determination of individual component operation within a system is needed. One must also balance the degree-of-difficulty required to access, program, and download data from these systems versus that of using dataloggers. On the other hand, some critical information monitored by the RMCS - refrigerant pressures at suction, discharge, and head for example - are quite difficult and expensive to measure due to the legalities of opening a refrigeration system to a pressure transducer. In this case, the use of the RMCS could prove quite useful. Each project should consider the balance of datalogger versus RMCS points based on the specific metering tasks and equipment at hand.

3.0 Historical Performance and Disaggregation of USAF Academy Commissary Energy Use

This section discusses historical energy use at the USAF Commissary, and also presents the results of submetering intended to illustrate where energy is going in the store by end-use function. This latter exercise is often referred to as ‘energy end-use disaggregation’. There are a number of reasons for disaggregating energy use within a facility. They include the following:

- To assist with calibrated computer modeling, using the measured values to correct models for actual building performance against known weather parameters
- To determine if energy use of various systems is in line with expected patterns
- To provide a baseline by which to measure future energy-related improvements

Energy Use and Maintenance Costs in Supermarkets

According to a recent DoE report, there are about 30,000 supermarkets in the U.S. which represent over 75% of the food sales market. The balance of the industry is represented by 102,000 small convenience stores and 11,500 small grocery stores (defined as doing $2 million or less per annum).

The installed cost of a typical 100 ton supermarket commercial refrigeration system is about $1 million, or about $10,000/ton. About 50% of this cost, or $5,000/ton, represents the cost of display cases. Maintenance costs for a parallel rack is about $0.75 per square foot of store sales area. Refrigeration maintenance costs constitute about 25% of a typical private- sector supermarket’s revenue. Another observation from the DoE report are that compressors and air-cooled condensers have an expected 10-year life (this is a manufacturers value; the in-situ life was not reported but is almost certainly lower).

The National Buildings Energy Consumption Survey (NBECS) indicates that the energy use intensity of a supermarket ranges from 175 to 410 kBtu/sf/yr, with a median of about 300 kBtu/sf/yr. Reports indicate that load shapes for supermarkets are relatively flat on an annual basis, with much of the load being driven by commercial refrigeration systems which operate with only a second-order effect on weather conditions. On a daily basis, there is an increase in energy usage during the day that can be attributed to occupant gain, increased lighting levels, food prep and office equipment, and some effect from weather due to infiltration, ventilation, and some increase in refrigeration load due to high condenser temperatures.

USAF Academy Utility Billing History

The monthly utility bills were collected and are provided in Table 3.1 below. Analysis and comparison of monthly readings is not possible due to inconsistent and unreported meter reading dates; i.e., all monthly readings below - while accurate - are not necessarily consistent with the month attributed. Only annual summaries are statistically valid. This is not considered a significant problem since commissary energy use is not strongly coupled to weather conditions.

Error! Not a valid link.

Table 3.1. USAF Academy Commissary: Utility Costs and Summaries

Unfortunately, a time history plot is rather meaningless for these monthly values since there is no indication of when the meters were read, and an interview with installation personnel indicated that these readings are accurate but not necessarily done at the same time of each month.

Overall, annual energy use at the USAF Academy commissary compares favorably with results of published national averages. Reasons for this are that this store is relatively new, the climate is favorable (moderate and low humidity), and the store itself has humidity control. Still, there are numerous opportunities for operating the store in a more energy efficient manner, as is discussed in Section 4 of this report.

Gross sales for FY94 were $23,190,000. Utility costs for the calendar year 1994 were $110,825. Civilian supermarkets indicate that utility costs are range from 0.85% to 1.5%, with a solid median at about 1.3% of gross annual sales. Direct comparison with DeCA facilities is not possible, since commissaries gross sales operate at a fixed 4% over cost.

Analysis of Flat Electrical Rate Charged to DeCA

Most military installations are on a demand-billing schedule, where separate charges are levied by the utility for energy used [$/kWh] and for the peak power required (demand) over some given time [$/kW peak]. A flat electrical rate is usually assessed by the host installation to the commissary and to other reimbursable customers on post. This is typically necessary because of the shortage in labor at the installations to read meters on a fixed schedule and the relatively low percentage of building meters capable of providing a demand reading. The relatively constant load shape of a commissary makes the use of a flat rate quite practical, provided the rate is set in a fair manner to both parties (i.e., DeCA and the host installation).

A flat rate of $0.04186/kWh is charged by the USAF Academy to the commissary. These flat rates are typically set by the installation Energy Manager and are usually an educated guess at best. The intent is to simplify billing of the commissary as a reimbursable customer. As of Jan 97, the installation was paying an on-peak cost of $0.047/kWh and an off-peak cost of $0.019/kWh. Definitions are:

On peak:	Apr-Sep	11A - 6P (M-F)	912 hrs	10.4%
	Oct-Mar	4P - 10P (M-F)	782 hrs	8.9%
Off-peak:		all other hours	7066 hrs	80.7%

(Holidays are off-peak, but not included)

An annual flat kWh rate (not incl. demand) can be obtained by:

($0.019/kWh)(0.807) + ($0.047/kWh)(0.193) = $0.0244.

Demand charge at the installation was quoted as $0.2042/kW/day. The installation set a peak of 14,800 kW in FY96. The highest peak set at the store during the June metering period was 365 kW, set during a short 2 minute interval; the highest peak over a 'sliding scale' 15 minute period (the worst possible 15-min peak) was 346 kW. Using a value of 350kW as representative, the demand charge for FY96 attributable to the commissary would be:

Demand: $0.2042/kW/day x 365days x 350 kW = $26,100/yr
Avg. On/Off Peak Energy charge: $0.0244/kWh x 2310144 = $56,400/yr

Total FY95 cost to installation of commissary operation was about: $82,500 /yr.

The commissary was billed: $96,564 /yr, which is 17% higher than the actual cost to the installation.

A more appropriate rate for the commissary should be about: $82,500/2,310,144 = $0.036 (vs. $0.0416)

<u>There is a potential for cost savings on the order 15-20 % through revising the flat rate energy charge being assessed by the Academy to the commissary.</u> This analysis attempts to be conservative, although it is limited by use of the peak set during the 3-week June metering period. If the commissary peak doesn't match the installation's peak, the correct flat rate would be lower than that stated above. Also, the rate structure used by the City of Colorado Springs (obtained from the Academy Energy Office; 719-333-4540) was in some question due to the need to better define any demand ratchet that may be in place. It should be noted that, in general, this type of billing error based on a 'best guess' flat rate could go either way, and that the current billing rate was based on a good faith estimate by the Academy energy staff.

Methodology for Disaggregation Measurements

The typical method for disaggregating end-use energy is to first examine the one-line power diagram (or the electrical panel layout in the absence of design documents). A metering plan is then designed to track power flows to various loads. When using dataloggers, the primary measurements are typically taken as a current using a split-core current transformer (CT) on each of the three phases at an electrical panel. Unlike the measurement of a balanced delta load such as a compressor or fan motor, each phase current requires separate measurement because the loads at the panel are wye-connected and are not typically in balance. The CTs used do not have the capability to measure power; the only quantity logged is current.

Power was calculated by multiplying the measured current times the voltage (which is known and does not vary much over time) and times a power factor. Power was also calculated by using a hand-held power analyzer (one that reads Volts, Amps, and Watts) to take a 'snapshot' of the circuit and determine the ratio between current and power (for example, suppose the power analyzer read 100 Amps and 20 kW on a circuit; then if the dataloggers recorded an average current of 130 Amps, the average power would be 130A / 100A * 20kW = 26 kW). The latter method gives a more accurate power for a specific circuit but does not account for variations of the power factor over time.

Voltage at the main panels was found to be about 268V and current was well balanced over the three phases. Overall power factor measurements at the main panel were noted to vary between 0.72 to 0.93 during installation of equipment, with a representative average of about 0.86. Power factor at the high voltage (primarily fluorescent lighting) was typically 0.97 or greater. These measurements were not logged, but are similar to logged measurements taken at the Fort Lewis Commissary and published by the Pacific Northwest Laboratories. The loads which contribute to low power factor tend to be electronic equipment on the low voltage side of the transformers (120/208V) such as office equipment and computer-based checkout equipment.

Results: Energy by End-Use at the USAF Academy Commissary

A breakout of electrical energy by end-use was obtained for the 21-day period by measuring 3-phase current at two-minute intervals at the respective electrical panels serving each load. Specifically, the following electrical panels were metered:

- Main power
- Lighting (included transformer to checkout stand equipment and other misc. loads)
- Refrigeration (compressor racks, condensers, defrost, case lights and fans)
- HVAC (fans, pumps, chiller)
- Miscellaneous/Other Equipment (differential between other measurements)

Further disaggregation is possible and relatively simple to perform, but was not attempted due to the greater interest in the diagnostics and performance testing of individual systems. A significant source of error is present in that the power values were taken using inexpensive current transducers and are not true RMS values. Combined with some voltage fluctuation, this makes the results only good to within +/- about 5%.

Table 3-2 shows the results of the disaggregation measurements. Total electrical power over the June 1996 period averaged 272 kW, as compared to the annual 282 kW (1994) and 264 kW (1995) average values. This can be taken to indicate that no major improvements or problems have occurred. Note that a comparison against the June measurements is not necessarily valid, since the utility meter readings cannot be associated to number of hours between readings. Results of energy by end-use are similar to those available from published results.

	% Ref.	% Lights and Other	% HVAC	% Other	
Academy	**57%**	**26%**	**13%**	**3%**	**Measured June 1996**
Wisconsin	56%	31%			
EPRI	54%	25%	17%	4%	Total Store Energy Management, Hussmann
Typical	51 - 67%	17% - 23%			FMI Conference, Younger

Table 3-2. Disaggregation Results (Baseline) and Comparison to Other Published Values.

4.0 Analysis of Academy Commissary Energy Systems

This section of the report details the findings and recommendations of the diagnostic and performance testing for the energy systems at the U.S. Air Force Academy commissary. For example, references to 'Figure 4-xx' refer to figures included in the text portion of the report; references to 'Figure C.x-xx' refer to figures and tables located in Appendix C.x.

Appendix C contains only a selection of plots which may be of interest in interpreting operation of various systems analyzed. A complete printout would be overwhelming and serve little purpose. Not all plots in Appendix C are referenced in this report, nor are all data streams present that are available or are calculated (see section 2.0 for a summary of all data available). For simplicity, some of these plots contain handwritten illustrations or explanations of the data being presented. All data collected is available from the authors and would be required for further investigation of specific system problems.

4.1. Main Air Handling Subsystem (DD-1) and Store Retail Zone

Physical Description: The constant-volume main air handling system (DD-1) consists of a 40 hp supply fan (19 kW measured), supply and return ductwork, a 4-row chilled water coil with pneumatically-actuated 3-way modulating control valve, a hot water coil with modulating valve, and a bank of heat reclaim coils which provide auxiliary heating using hot refrigerant gases from refrigeration racks 2, 3, and 4. Chilled water coil capacity is 415,000 Btu/h and face area is 38.5 sq. ft. (35 ton nominal); hot water coil capacity is 850,000 Btu/h and 38.5 sq. ft.; and heat reclaim coil capacity is 716,000 Btu/h and 30.5 sq. ft. Return air is taken from a tunnel system below the cases to remove unwanted cold air from the refrigerated aisles. A Munters SuperAire S30 desiccant system is provided to dehumidify air taken from and returned to the mixed air chamber via a 5 hp process fan (2.8 kW measured). Dehumidified air processed by the system is thus a combination of outside and return air. A motorized damper controls outside air flow between the mixed air chamber and the desiccant system. Reactivation of the desiccant system is achieved by a heat reclaim coil (105,600 Btu/h design) from low temperature refrigeration Rack 1, followed by a 2-stage natural gas-fired in-duct furnace (400,000 Btu/h in; 308,000 Btu/h out). During desiccant operation, aftercool is supplied by the chilled water coil. Reactivation air consists of 100% outside air provided by a 7.5 hp draw-through reactivation air fan (3.9 kW measured). A small motor rotates the honeycombed desiccant wheel at 8 revolutions per hour when the desiccant system is activated. A Danfoss NC-25 RMCS is used to control HVAC system operation.

Metering Description: A schematic of DD-1 subsystem is shown in Figure 4-1.

\

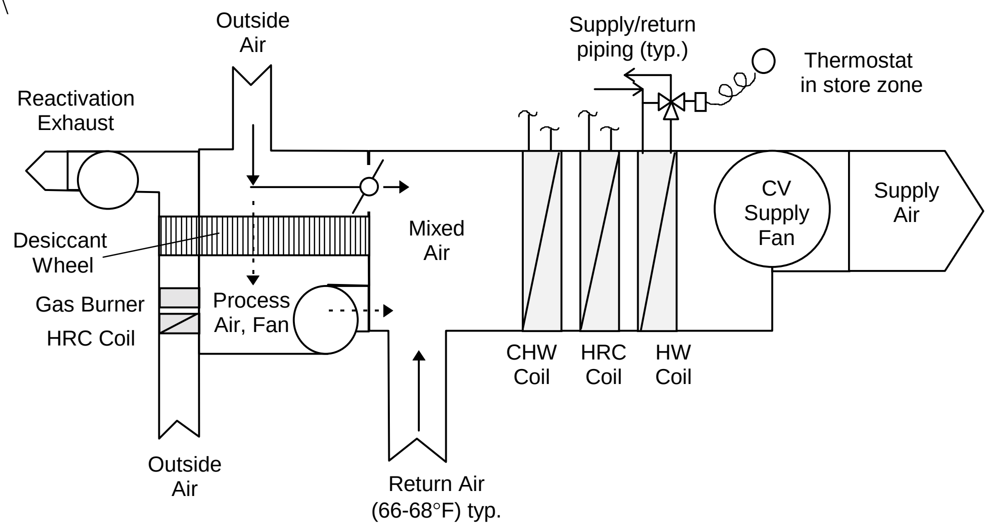

Figure 4-1. Schematic of main store air handler w/ desiccant system

Data collection points were taken as follows:

- Supply air temperature and relative humidity (RH)
- Return air temperature and RH
- Mixed air chamber - average temperature of 2 sensors
- Processed (dehumidified) air temperature and RH
- Current to supply air fan (surrogate for flow)
- Current to process air fan (surrogate for flow)
- Current to desiccant wheel drive motor (surrogate for operation)
- Current to reactivation air induced draft fan (surrogate for flow)
- 'Cold' store zone temperature and RH (near refrigerated aisle)
- 'Dry' store zone temperature and RH (non-refrigerated aisle)

DD-1 Observed System Performance: Zone DD-1 was controlled to maintain a 72 to 73°F setpoint 24 hours per day throughout the metering period. A humidistat also was shown to maintain a maximum of 45% RH in the zone (45% zone RH - on; 40% zone RH off) in accordance with DeCA directives. Data in Figures C.2-1 and C.2-2 shows a relatively constant supply air temperature of about 72°F (with variance between 67 and 77°F minimum and maximum), and resultant temperature changes across the coil required to maintain space conditions. Analysis of this data also indicate a store 'balance point' of about 80°F outside air, below which the zone is in heating mode, and above which the zone is being cooled. Modulating valves on the hot and chilled water coils are controlled from zone temperature as follows:

68°F: 100% hot water	73°F: 0% chilled water
72°F: 0% hot water	77°F: 100% chilled water

Supply air temperature is thus variable, and is a function of % valve setting and % OA (assuming supply air CFM, return air temperature, and hot and chilled water supply temperatures are relatively constant). This control strategy results in rather tight control of store temperature and humidity. Through the period, store air temperature at the datalogger location varied from a value of 75°F maximum to a minimum 69°F. The above control strategy and resultant supply air temperature indicates that the flow rate of supply air far exceeds that needed for space temperature, humidity control, and ventilation requirements. Data taken during the June period indicate that the air-side of this system is in control, although improvements can be made to the control strategy (see 'Deficiencies and Recommendations' below).

The supply air fan is operated 24 hr/day regardless of zone needs. The assumed purpose of this commonly used supermarket schedule is to prevent temperature stratification between dry grocery and cold aisles within the zone. A testing and balancing (TAB) report dated at the June 1992 system startup indicated a measured supply air flow of 26,500 CFM (about 1.0 CFM/sf). This is in accordance with ASHRAE recommendations for supermarket design, and is also very close to the design 26,000 CFM indicated on the drawings. However, measured power at the fan motor indicate only a 19 kW constant power draw and the given fan/motor system should be fully loaded at about 30 kW to provide the stated CFM. At the current time, the fan is only about 63% loaded and is most likely not providing design airflow. Also contributing to the problem is slightly excessive negative pressure on the suction side of the fan (-2.5 in w.c.). The source of this problem could not be positively identified, but restricted flow on the return side of the system is suspected, probably at the intake slots at the kickplate below the freezer cases. The return air tunnel system was inspected and showed no apparent obstructions or restrictions.

The 'economizer plot' (Figure C.2-19) indicates a constant 40% outside air fraction being drawn by the system. Minimum design OA requirements for ventilation purposes are indicated on drawings to be 2,025 CFM. The motorized OA damper was disabled and a bucket wedged into the damper to force maximum OA at all times. According to maintenance personnel, this was done to 'provide adequate fresh air' and to try to alleviate the excessive negative pressure at the fan.

Due to low ambient humidity at the Academy, the desiccant system is rarely fired. However, during those times that the desiccant portion of the DD-1 system is activated, there is a problem with fan sequencing. The problem is most likely originating in the control logic (Mr. Carlos Delarosa, 800-229-8557). Munters technical literature states that at RH setpoint, the process fan, reactivation fan, reactivation furnace, and cassette (wheel) motor should activate simultaneously. When the minimum RH point is satisfied, the process fan and burner shut down, and a user-defined 2 - 10 minute thermal purge causes the reactivation fan and wheel motor to continue. Time-series data in Figures C.2-

3 through 7 show that the process fan is operating in short cycles, while the reactivation fan and wheel motor are operating at much longer intervals. Determination of which system is in error (process or reactivation) is difficult to determine because of the use of a separate data point to surrogate the zone humidity, and due to poor calibration of the humidity sensor in the zone. The system is controlled by store ambient RH. Data show that the 45% maximum setpoint is controlled well by the system, although the reason for 'high' store humidity (in excess of 40%) warranted further investigation. Analysis of the overall operation yields the conclusion that there is no good correlation with either temperature or RH, but that the desiccant system operates when humidity ratio rises above about 0.018 lbm water/lbm dry air. At a firing rate of 400,000 Btu/h (at sea level; about 4 therms/hour) and with 6.7 kW total fan power, cost of running the air-side of this system is less than $2/hr. This neglects the energy cost to recondition warm air introduced into the mixed air chamber. It also neglects the fact that heat reclaim could supply up to 40% of the thermal energy required for reactivation of the desiccant wheel (the heat reclaim system using hot refrigerant gas from rack 1 was not operational). Although humidity control via the desiccant system is relatively inexpensive once a system is in place, high humidity is best controlled first through judicious use of outside air and supply air to the zone, followed by use of a desiccant system (or 'dual-path' vapor compression system located at outside air intake only) to take up the difference.

During a follow-up visit in September, it was noted that there were a number of changes from the previous data gathering period. There was no heating available to the store due to the hot water being valved off manually at the DD-1 supply coil. Chilled water - originally valve-balanced at the circulation pump to about 50% flow - was put back to 100% flow. Discussion with a Scott-Polar technician indicated that this was due to reducing the temperature difference across the chiller from ΔT = 12 °F to ΔT = 8°F (see 'Chiller System' section below). The system was tested through its limited control strategies by applying both heat and humidity to the thermostat in the zone to determine if there were problems with the heating and/or dehumidification modes. Both modes operated as scheduled, although the modulating valve on the heating coil had to be readjusted to allow for proper full range operation. Since cold weather was expected and the store calls for heating below OA = 80°F, the hot water supply valve was returned to its normal position.

Also during the follow-up visit, an experiment was performed to determine the temperature stratification which would result if the supply air fan were turned off during store off-hours. The air handling system was shut down at 1800 hrs on Sep 4 and reactivated 13 hours later at 0730 hrs on Sep 5. Temperature and humidity were measured in the zone in both the dry grocery and in the cold section, as well as at outdoor ambient and supply and return duct locations. Results of zone temperatures are shown in Figure 4-2. A slight stratification between dry and cold areas set up rather quickly until a new equilibrium condition was reached. Contributing to comfort conditions overnight was the absence of cool outside air into the store, which would need to be heated during a typical night operational mode. Discussions with night personnel indicated that nobody noticed any change in space conditions overnight. A sharper temperature stratification was also observed to set up vertically at the cold aisle temperature/humidity station. This lower temperature near the floor (about 57°F at 6 in. above floor level) could cause discomfort for overnight stocking personnel only if they were to insist on wearing shorts or sandals while they work.

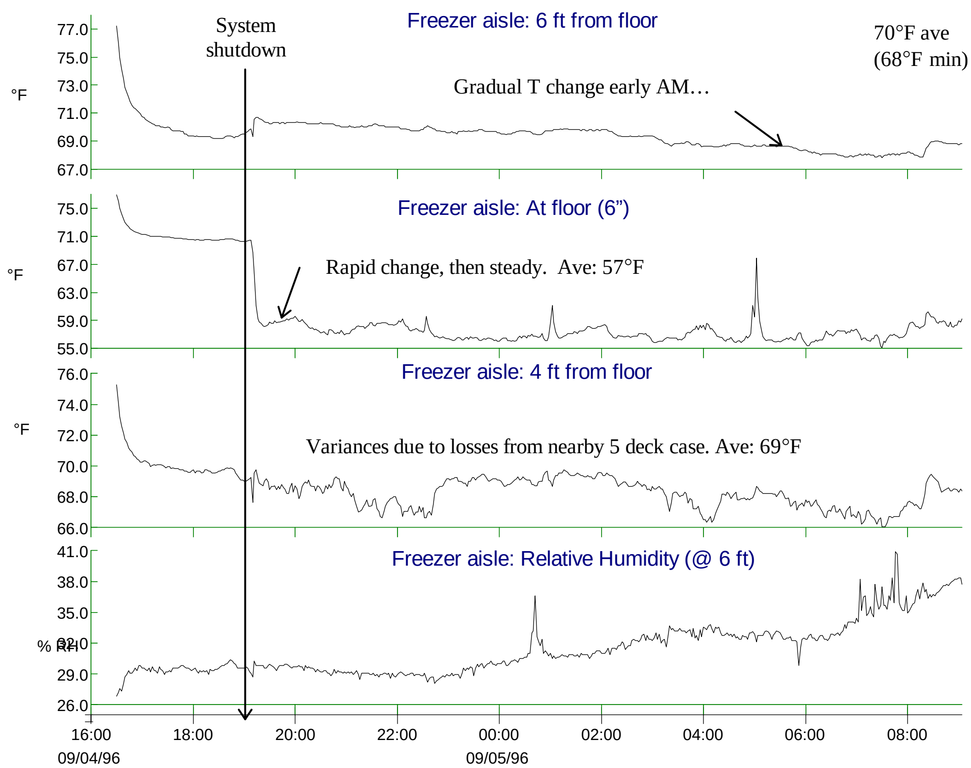

Figure 4-2. Zone Temperature Results of Night Setback Experiment

Single measurements of CO_2 were taken as a surrogate for indoor air quality concerns. There was no attempt to operate ventilation fans I other parts of the store to provide fresh air during this experiment, although that may be a preferable The results are as follows:

Time (hour)	CO_2 level	Notes
1800	730 ppm	System Shutdown at 1815
2100	710 ppm	
0800	760 ppm	System Restarted at 0815

<u>DD-1 Deficiencies and Recommendations:</u>

- A full pressure testing and balancing study of all air-handling systems and exhaust systems should be implemented to determine actual flow rates for the existing modes of operation. This study should be implemented prior to initiation of any setback strategies discussed below. The supply air fan is about 60% loaded, and is operating inefficiently. Also, pressure differences resulting in unwanted interzone flow between the main floor, loading area, and walk-in boxes should be checked out prior to any further changes being implemented.

- Reduce supply air CFM. Currently, store CFM is designed for and supplying at or slightly above 1.0 CFM/ft^2. This is in accordance with published ASHRAE rules-of-thumb. However, EPRI research indicates that design CFM requirements for stores with humidity treatment of outside air (via desiccant or dual path technologies) should be in the 0.5 to 0.7 CFM range, assuming adequate OA can be delivered within these parameters. This observation is backed up by the fact that the store temperature setpoint of 72°F is being maintained with supply air of only 70 - 75°F over a wide range of OA conditions. This indicates that there is excessive mass flow being supplied by the air system. Reduction in supply CFM would result in lower fan energy use, increased supply air temperatures during heating mode, and decreased supply air temperatures during cooling mode. This behavior can be seen at the bottom of Figure C.2-1 showing the temperature increase and decrease across the coil. Savings could also be obtained if either the fan system were cycled during store hours when there is little load on the zone (those hours near the balance point). There is a need for some minimum constant flow during summer operation, since customers in summer clothing may find the cool temperatures that can set up near the floor objectionable. While a short experiment performed during off hours found the problem to be quite tolerable, further experimental or calibrated modeling is needed to determine the allowable 'off-time' of the fan system and the temperature-time response of the sub-zones to supply air changes.

- Implement night setback of the main floor zone. Continuous operation of the 20 kW main supply air fan results in unnecessary energy use by the fan itself and by the heating and cooling supply systems. Limiting fan operation to store hours only plus 2 hours lead startup prior to opening (2964 hrs annually) reduces fan electrical consumption from $7,400/yr to about $2,500, a savings of $4,900/yr on supply fan energy alone. Heating energy savings are not included in this figure, and it may be assumed that cooling energy use at night by the zone is insignificant. A minimum zone temperature of 55° to 60°F is recommended for comfort of night stocking personnel In the event that the zone decreased to the night setback, the system should override setback to maintain minimum conditions. It is also possible to control the zone using CO_2 measurements, although it is not apparent that high CO_2 concentrations would exist in the store as a result of night setback. Additional energy savings on the refrigeration system would also be obtained due to the lower store ambient temperature near the cases.

- DD-1 outside air damper is not functional (40% fixed OA fraction). Proper control of the outside air dampers would result in reduced energy use from both the hot and chilled water supply systems. Also, there is no ‘free energy’ being used by the building when available from outside air. Control of OA should account for a minimum ventilation standard at point where heat reclaim can no longer supply load. OA damper might be controlled on enthalpy due to availability of low moisture ambient air at 7,200 feet elevation.

- Heat reclaim (to supply space heating) is not functional. Observations and measurements indicate that all space heating needs are provided by hot water supply. Investigate reason for heat reclaim not functioning in the event that there were prior operational problems on the refrigeration system. If none are indicated, heat reclaim should be reinitiated. The store is being heated whenever outside air is less than 80°F, which according to weather data is only about 480 hours per year (less than 6% of year).

- Desiccant function on DD-1 should be fixed. Possible source of problem is air flow switch 1AFS. This switch is used as proof of flow for desiccant system. Pneumatic tube connected to low pressure tap may be severed. Humidistat in zone needs to be calibrated, and fan sequencing needs to be reconfigured. Although the initial cost of a desiccant system is probably not justified at this location, the fact that it is installed, is an integral part of the air handler, and is inexpensive to operate makes the relatively low cost of recommissioning the system justified. Also, the use of a lower setpoint (35%) should be investigated. This RH often exists naturally in the zone, although the savings to the refrigeration system due to a change in setpoint from 40% to 35% is less than that from the 55% design RH to a comparably lower value. Modeling of the dehumidification effect on the refrigeration system can be done by the FEDS software using the commissary building set, or could be done using the EPRI Supermarket Application Program (SAP) when it is made available.

- Fire damper actuator linkages were found disabled on DD-1 supply and return air ducts. Investigate reasons, repair accordingly, and insure fire alarm and suppression system is functioning properly. No energy savings associated with this deficiency.

- Increase deadband on heating and cooling of DD-1 zone. There is a 1° deadband from 72°F to 73°F, above and below which cooling and heating are implemented respectively. This deadband could be increased to about 2 or 3°F. However, if a continuous supply fan operation strategy is continued (i.e. the fan is not cycled) the only energy savings available will be in the slightly increased efficiency associated with longer runtimes of cooling and heating equipment. A more compelling reason for this change may be in the decreased life that occurs due to the excessive equipment cycling observed. This issue is further addressed in the heating and cooling supply systems section below.

- Identify and correct problem with store zone lighting control. Controls which lower lighting levels during off-hours was never properly installed or commissioned and was eventually overridden locally. This is a commonly employed strategy and should be re-implemented if possible.

4.2 AHU-2 (Administrative Area Air Handling Subsystem)

Physical Description: The roughly 3,000 square foot administrative office space is served by a constant-volume air handling system AHU-2. This system is supplied by the primary chilled water and hot water supply which are connected to the packaged chiller and boilers. This unit has a fixed outside air intake, and simultaneous measurement of supply, return, and outside air indicate a fixed 25% outside air fraction. A 1 hp (0.6 kW measured) supply fan provides 1570 CFM design air flow to the administrative zone. A thermostat with setback feature was located in the zone, but a setback schedule was not being implemented.

Metering Description: A typical constant volume air handler metering plan was employed for this system. It was determined that zone temperature was approximately equal to the return air duct temperature. Data collection points for AHU-2 were taken as follows:

- Supply air temperature
- Return air temperature (also surrogate for zone temperature)
- Mixed air temperature
- Current to supply air fan (surrogate for flow)

AHU-2 Observed System Performance: Figure 4-3 provides a time series overview of the admin zone performance. The supply air fan (not shown) ran continuously throughout the period and was set to 'hand' operation. With a fixed 25% OA fraction, this continuous air stream requires conditioning at almost all times. Temperature histories in the zone during the June period (using return air) show interior temperatures 'profile-tracking' OA, with zone temperatures exceeding 74°F as OA exceeds about 85°F (see Figure C.4-1 through C.2-4). A more typical setpoint-deadband variance is not visible for this zone. There are two possible reasons for this behavior; either (i) the return air is not a good surrogate for zone temperatures (due to stratification), or (ii) there is a problem with control of the chilled water valve at the AH-2 coil or there is insufficient flow at that coil. Stratification may be the cause, but this issue should be investigated further since discussion with the management and office staff revealed some historical degree of dissatisfaction with the control of temperature in this zone, particularly in summer months. One intermittent anomaly can be seen in Figure 4-3 shortly after midnight each day for about one week (6/13 to 6/19). The supply air temperature would increase to about 100°F (corresponding zone temperatures reached over 78°F) for about 30 minutes. The zone would regain its setpoint by 0100 hrs. No reason for this behavior was apparent, although a night setback thermostat located in the zone is suspect.

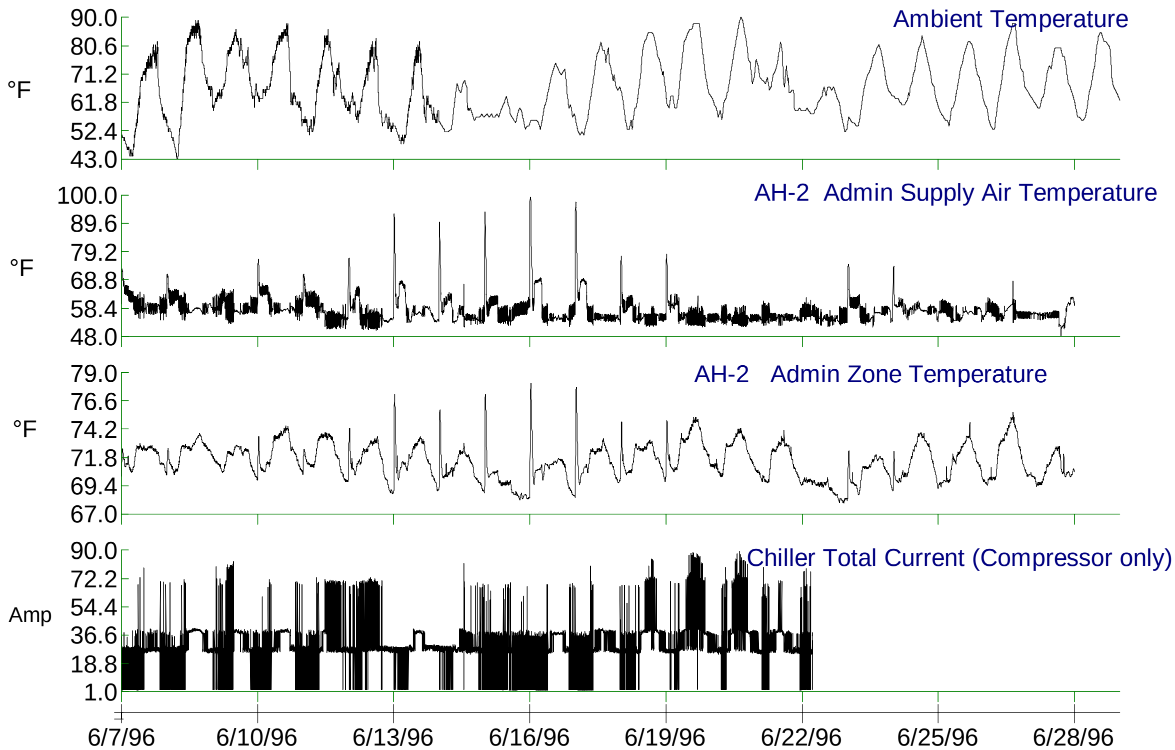

Figure 4-3. Overview of Admin Zone Performance versus Time

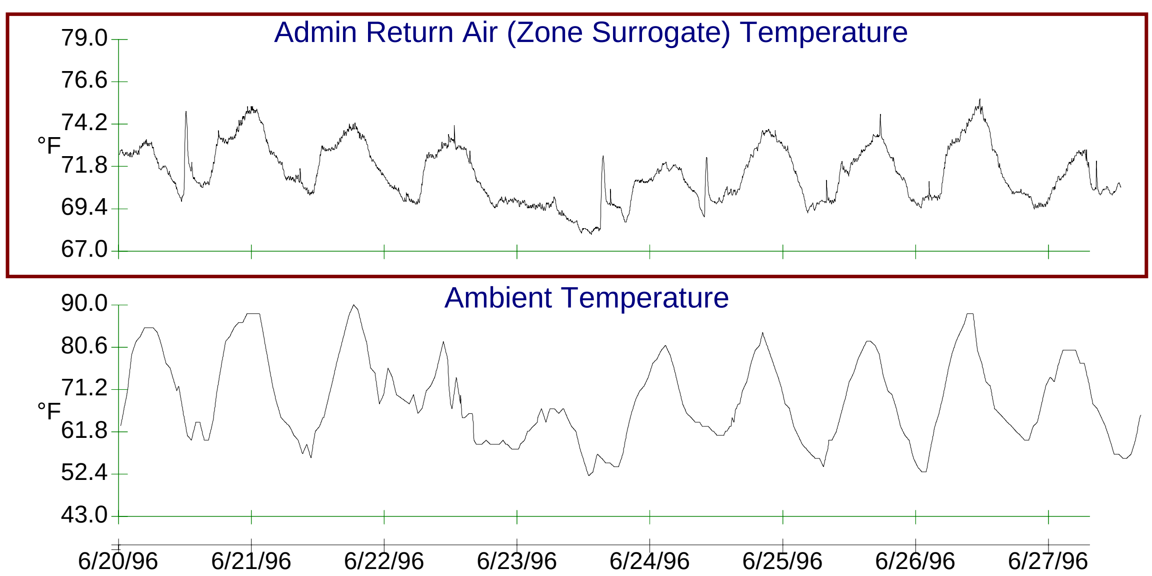

Figure 4-4. Problem: Return Air (Zone Temp) Tracking OA

AHU-2 Deficiencies and Recommendations:

- Implement night/off-hour setback (restore fan operation to 'auto' and setback temperature) on admin zone (AH-2).

- Implement economizer control on AH-2. There is good opportunity for cooling office space with ambient air at this location. More traditional use of economizer.

- Consider use of a dedicated DX unit for this zone. This relatively small load is currently supplied by an oversized packaged chiller unit (see 'Chilled Water Supply Subsystem', below), and implementation of night setback could further reduce the load on this unit. Benefits of adding a dedicated DX unit include:
  a. Greatly improved temperature control in zone.
  b. Better control of packaged chiller (allows more hours of chiller lockout).
  c. Improved energy efficiency (better load matching, no distribution losses, economizer operation).
  d. Reduced service calls.

4.3 Chilled Water Supply Subsystem (Packaged chiller and distribution)

Physical Description: The chilled water supply subsystem consists of a Dunham-Bush 65-ton, air-cooled packaged chiller unit using R-22 refrigerant, 2- 7.5 hp chilled water circulation pumps (4.7 kW each measured), a 6-fan air cooled condenser located on the rooftop, and associated plumbing, piping and valves. Zones served by this chilled water loop include the main store retail zone (DD-1), bakery (AH-1), admin area (AH-2), and 6 small fan coils located in small office space around the store. The chiller unit has two reciprocating Copeland Discus compressors (CH1 and CH2) rated at 30 and 35 hp respectively (23.4 and 27.4 kW measured). Chiller control calls for the following sequence of operation:

Sequence	Description of compressor operation	% capacity	tons capacity
Mode 0	Both off	0%	0
Mode 1	Compressor 1 on (50% loaded)	23%	15
Mode 2	Compressor 1 on (fully loaded)	46%	30
Mode 3	Compressor 2 on; Compressor 1 on (50% loaded)	77%	50
Mode 4	Both compressors full on	100%	65

Metering Description: Data collection points were taken as follows:

- Chilled water return temperature (entering water temperature, EWT)
- Chilled water supply (leaving water temperature, LWT)
- Current to packaged chiller compressor 1
- Current to packaged chiller compressor 2
- Current to chilled water circulation pump CP1
- Current to chilled water circulation pump CP2

Chilled Water Supply - Observed System Performance: The chiller and chilled water supply subsystem is oversized for the application and location. Inspection of the design documents indicates that a 50 ton chiller was specified for the project, but a 65 ton unit appears to have been substituted during construction. No apparent reason for this is indicated. The measured balance point for the main store zone is about 80°F, below which little or no cooling is needed for the majority of the building. There are two other major zones on this chiller - the admin office served by AH-2 and the bakery served by AH-1 (not analyzed). Load schedules and thermal behavior of these zones are so different from the store zone that a better design solution would have been to supply admin and bakery via their own small rooftop DX units.

Most problems observed during the period can be directly traced to the oversized chiller and chilled water distribution system. Excessive cycling was first observed during the day the metering equipment was being installed. Eventually, the chiller alarmed out, and a Scott-Polar technician was called. No apparent reason for the alarm was found, but a datalogger was installed overnight with a 3 second sampling rate to help Scott-Polar diagnose the problem. Figures C.5-1 through C.5-4 show the results. The chiller had periods where it would cycle on, proceed rapidly through its various modes of operation, then power down. Cycles consisted of about 7 minutes on, then 7 (or less) minutes off. Not apparent from the plots was the fact that there was a consistent attempt by compressor 2 to start, followed by an instant shutdown. Since the entire sequence was less than 3 seconds, only a few of these can be seen sporadically in the 3-second samples. This data was sent to Scott-Polar for further action. A follow-up call to Mr. Dave Moore (contract manager; 801-973-5930) indicated the diagnosis was a control problem, and that water temperature setpoints were changed to alleviate it. While the acute problem was solved in this instance, the chronic problem still persists due to oversizing.

The chilled water circulation pumps are also oversized. A manual valve at the pump head was throttled down to about 50% position and locked in place during the two initial site visits. Discussion with the original contractor (Hussmann) indicated that the pumps and supply were in fact too large, and that the throttling was an attempt to get the system into better balance and control. The contractor also indicated that this had been in place since shortly after building occupation with no problems. However, sometime during the monitoring period, this valve was returned to the full open position. Maintenance records (now under Scott-Polar) explained that this was done in response to a 'hot-call' from the admin area, and the valve was opened in an attempt to get more chilled water to the coil.

In addition to the maintenance costs and poor performance attributable to excessive compressor cycling at the chiller, there is an energy cost due to the time of operation during cycles. Analysis of Figures C.5-5 and C.5-6 indicates that the total area under the curve (in amp-hrs) is equivalent to 8442 kWh over the 14 day period which was analyzed. An estimate of the area under the curve during periods of low cycling provides a value of 4345 kWh. This illustrates that an energy reduction of 50% (equal to a cost of about $350/month, or 4% of monthly cost) could be achieved in addition to the O&M benefits of reducing cycling.

<u>Chilled Water Supply - Deficiencies and Recommendations:</u>

- Implement OA lockout on pumps. Pumps should be interlocked with boilers to shut down power to above some temperature to be determined (about 70°F). This should take into account shutdown of each boiler individually, so that one boiler can be brought up as needed, and should also allow for changing the schedule of lead/lag boiler identification. This will help to insure approximately equal annual runtimes on each boiler.

- Investigate strategies for sustaining operation of oversized chiller. Representatives for the manufacturer (Dunham- Bush) suggested consulting with their applications engineers to find a short term solution that would alleviate the cycling problem. One strategy implemented by the contractor at USACERL's recommendation was to disconnect the second compressor from automatic operation. This action had occurred prior to our September 1996 follow-up visit. Data is not available on the performance. At the least, USACERL recommends a follow-up check by a Dunham-Bush rep.

- Chilled water pumps and motors oversized - trim impeller or replace motor/pump assemblies.

- Packaged chiller unit is oversized - often is serving just admin space. Potential for rescheduling chiller operation. Also, investigate potential for separate system. (This is a design issue - the two zones have quite different thermal behavior and are not well served by a single packaged unit, particularly one that may be oversized to begin with.

4.4 Hot Water/Space Heating Supply Subsystem

Physical Description: Two identical gas-fired Lochinvar packaged hot water boilers are used to supply hot water for all space heating needs. Loads include the DD-1 and AHU-2 air handlers discussed above, the bakery air handler (AHU-1), and 21 assorted unit heaters located throughout the warehouse and non-retail areas. Boiler capacities are 1,000,000 Btu/h input sea-level rating (800,000 Btu/h @ 7,000 ft), and 800,000 Btu/h output (rated 704,000 Btu/h @ 7,000 ft). The boilers are constant output designs, i.e., they are not designed for two-stage or modulating capacity. The two boiler system, however, is designed for staged operation so that system can fire at only 1.0 or 2.0 Mbtu/h. Two 7.5 hp (4.8 and 5.1 kW measured) centrifugal pumps supply hot water to the main hot water supply loop. Boilers are plumbed in parallel to a single supply pipe, which then splits to the two parallel pumps. Pumps are rated for 167 gpm @ 99 ft. head. Total measured hot water supply flow was constant at about 138 gpm during the June monitoring period. This value, measured via ultrasonic flow meter, shows good agreement with the 133 gpm value derived from temperature rise of water flow (calculations shown in Figure C.6-3).

Metering Description: Data collection points were taken as follows:

- Boiler 1 hot water return (inlet to boiler)
- Boiler 1 hot water supply (outlet to boiler)
- Boiler 2 hot water return (inlet to boiler)
- Boiler 2 hot water supply (outlet to boiler)
- Current to circulation pump for boiler 1
- Current to circulation pump for boiler 2

Hot Water Supply - Observed System Performance: Both boilers and pumps were on during the entire course of the June measurement period. With a balance point in the main store zone of about 80°F, some heating is in fact required by the store to maintain setpoint. However, most of this heating load could be met by use of the heat reclaim system (which was not functional), or by the use of only one boiler throughout the summer months. All hand and flow balancing valves were found in the full open position. Both pumps were set to ‘Hand’ operation. Hot water temperatures were consistent at about 170°F regardless of ambient conditions (there was no provision for hot water temperature reset). Boiler B-2 was responsible for heating during the measurement period, with a consistent setpoint of 170°F and a deadband of +/- 6°F. Further analysis shows that the boiler cycled 3 times per hour average - independent of ambient temperature - for 6 to 8 minutes per cycle (about 20 min/hr, or 33% duty cycle for one boiler; or 230,000 Btu/hr average boiler output to water, or about 56 therms/day x $0.338/therm x 30 days = $570/mo).

Hot Water Supply - Deficiencies and Recommendations:

- Implement OA lockout on pumps. No automatic load control present, resulting in both boilers on continuously. Pumps should be interlocked with boilers to shut down power to above some temperature to be determined (about 70°F). This should take into account shutdown of each boiler individually, so that one boiler can be brought up as needed, and should also allow for changing the schedule of lead/lag boiler identification. This will help to insure approximately equal annual runtimes on each boiler.

- Implement hot water reset with OA. 170°F HWS temperatures are excessive for a supermarket during daytime spring conditions, and may be high for summer evening operation as well. This correction would require the installation of a valve, actuator, and bypass piping to implement.

- Correct problem observed at boiler B-2 exhaust. Excessive corrosion is present on the top of a metal roof located directly below the exhaust of B-2. Problem most likely due to acidic stack condensation leaking onto roof top.

- Consider use of staged boilers upon need for boiler replacement. The current boilers at 1.0 million Btu/h are not capable of staged or 'high/low' fire operation. There are thus only two modes for the heating system - 1.0 or 2.0 million Btu/h firing rates.

- Consider impeller trimming on HW pumps. Manual shutoff valves being used to restrict water flow.

4.5 Refrigeration System

Physical Description: (See description provided in *Chapter 2.0* above).

Metering Description: Datalogger points were taken as follows (all points are single-leg current measurements from a 3-phase power supply and are used as surrogates for operation status):

- Rack 1 Compressors
- Rack 1 Defrost (single channel)
- Rack 2 Compressors
- Rack 2 Defrost (single channel)
- Rack 3 Compressors
- Rack 3 Defrost (two channel)
- Rack 4 Compressors
- Rack 4 Defrost (two channel)

Refrigeration System - Observed Performance:

Compressor Rack Performance

Pre-engineered, packaged refrigeration systems are typically designed, built, and installed properly. Maintenance of these systems varies, but is kept in relatively good repair due to the critical nature of the load and the necessity of an alarm when any disruption in performance is detected. A discussion of the overall energy characteristics and load shapes of the refrigeration subsystem is provided in *Chapter 2.0*. Other notes of interest can be seen in Appendix C.7 throughout the section on refrigeration. Direct measurement of rack performance (as a COP or kW/ton value) are extremely difficult due to (i) the need to accurately measure refrigerant flow rates; and (ii) the variable nature of the load and the use of a liquid receiver to compensate for the variable load. On way of normalizing rack performance is to use some easily obtained value - such as the design load in tons of refrigeration (TR). Using the areas under the curve in Figure C.7-1 for 3 days, the following parameters can be shown:

Parallel Rack	kWh/day	Avg. kW	Design tons	kW/ton*
Rack 1	770	32.0	13.4	2.4
Rack 2	520	21.7	10.5	2.1
Rack 3	529	22.0	27.5	0.8
Rack 4	N/A	N/A	37.5	N/A

- Compressor power only is included. This value was calculated using design case conditions (75°F, 55% RH). Actual store conditions were about 72°F, and 28% RH. Value does not reflect actual measured tons refrigeration, which would be about 30% higher under ideal operating conditions.

Although these racks are pre-designed and set up to handle the various loads imposed on them by the cases and walk-in circuits, a number of parameters can cause disruptive operation of the rack system. These parameters include varying store ambient conditions (air temperature, ambient surface/mean radiant temperatures, relative humidity, and store lighting), case loading practices, and walk-in cooler/freezer operation. Acute problems such as high head pressure limit cutout, and oil fault conditions set alarms and are addressed as they occur. Often, there is no readily identifiable source for these problems and the equipment is reset. In these instances, only when a problem persists is a source identified and corrected. Discussion with contractors throughout the supermarket industry indicate that the problem is due to a lack of time to correctly identify the problem. This lack of time can be due to a busy repair schedule or due to the lack of financial incentive to get to the source of a problem. These systems are complex, and proper diagnosis of a problem could involve analysis of the load side (cases and coolers), the compressor rack, the electronic control system, and/or the condenser subsystem. Many problems are therefore solved by trial-and-error after a number of service calls. In some instances the service calls are not all logged by the maintenance personnel.

One common chronic problem with compressor systems involves excessive cycling of one or more compressors in a rack. ‘Excessive cycling’ is a relative term. It does not necessary cause large amounts of energy use directly, but has a substantial impact on equipment life and performance, is a very inefficient use of energy, and may contribute to unnecessary demand charges. There are a number of sources for cycling problems. Design problems exist when a rack is specified that is not properly matched to the total circuit load imposed upon it. The most common source of excessive cycling is due to a leaking solenoid/control valve. Low refrigerant or a mismatch in circuit suction pressure (case temperature) setpoints can also cause cycling problems in an uneven parallel compressor rack. This situation also has inefficiencies due to the need to set evaporator pressure regulators (EPRs) to impose pressure drops in higher temperature circuits to match the lowest pressure circuit in the manifold. Often, there are load conditions on the rack which will cause unstable compressor performance (i.e., excessive cycling of one or more compressors) as the system moves through its varying loads, but these conditions should be transient in nature as the rack system moves from one stable equilibrium point to another.

The most consistent definition of excessive compressor cycling seems to be over 120 cycles per day. Most compressor manufacturers state that compressor life is defined by 750,000 starts, although factory design testing requires 1,000,000 starts under bench conditions of repeated starts with low runtime. Refrigeration technicians and system designer/manufacturer often disagree on a field value, the former feeling more comfortable with cycling in excess of say 100 to 200 times per day than the latter. However, technicians vary widely on their attitudes. An application engineer at Tyler Refrigeration Corp. stated that anything in excess of 60 cycles per day, consistently over a week or so, is cause for concern and should be looked into.

Figure 4-5 below illustrates a slight but acceptable instability from one load state at 0730 through an unstable period from 0915 to 1000, followed by stable load performance at 1030. Rack 1 serves the lowest temperature loads (ice cream and frozen foods) and uses a satellite compressor to lower the suction pressure for frozen foods to serve the lower temperature ice cream circuits. The top plot combines the energy use (amps @ 460V) of the first two compressors; the middle plot is the third compressor; the bottom is the fourth. In this case, the cycling occurs as a 'swapping' between compressors 2 and 3 to maintain a constant suction pressure in the common manifold.

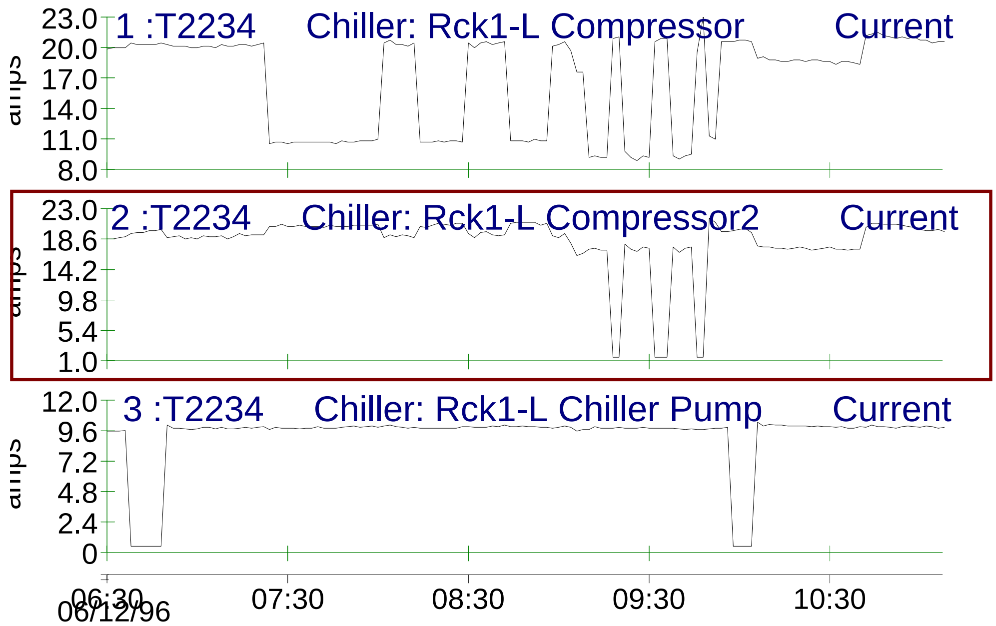

Figure 4-5. Acceptable compressor performance - load increasing from 0800 to 1030 (note compressor 'swapping' at 0930)

Figure 4-6 below illustrates a situation on Rack 2 where some changes to the suction pressure setpoint (SSP) or to the compressor control sequence may be warranted. This cycle (shown for the date 8 June 96) repeats daily for the entire 3-week period. The first three plots illustrate behavior of Compressors 1, 2, and 3 respectively (ignore labels to the contrary); the fourth is total rack amp draw (the sum of the first three plots). The cycling observed at compressor 3 is perhaps on the margin of acceptability, averaging between 4 to 5 cycles per hour, although it occurs steadily throughout the entire monitoring period. Overall, however, the rack could be considered rather unstable due to the fact that both compressors 1 and 3 are cycling, exhibiting a somewhat choppy behavior. A better compressor sequencing strategy or suction pressure setpoint modification should be able to smooth out this power signature. Also, it is possible that a solenoid valve is leaky, allowing refrigerant into the manifold that needs to be pumped out. Discussion with a Scott-Polar technician indicated that they would see these problems only after they have caused some alarm-setting type event, and even then may not know the root cause of the problem.

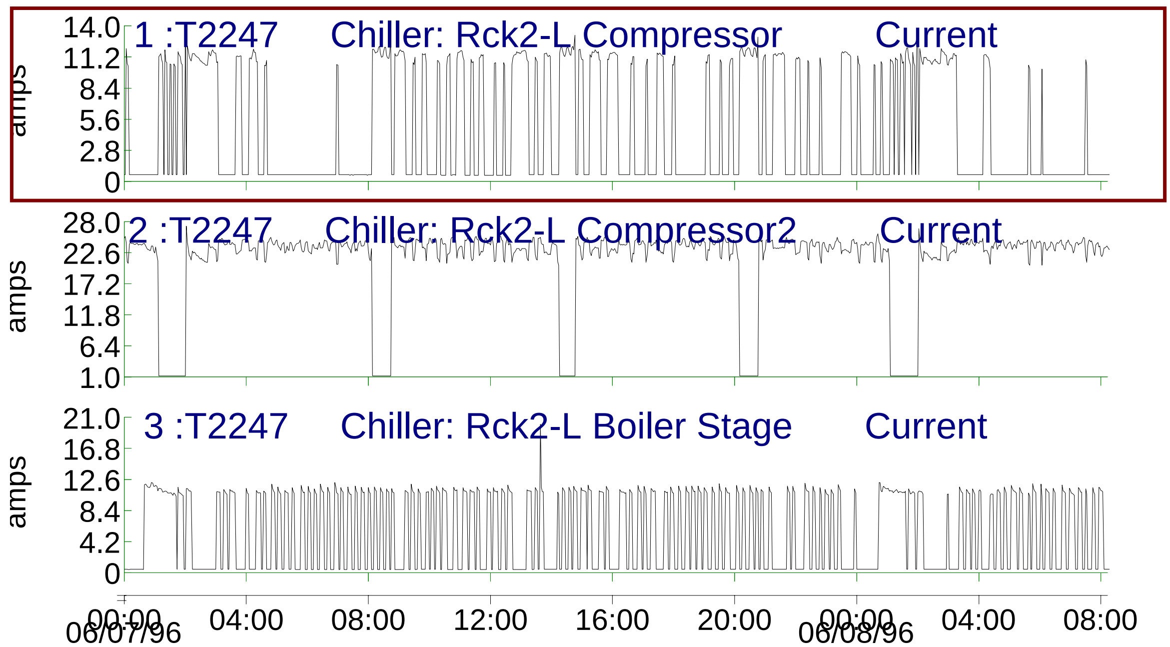

Figure 4-6. Rack 2 Compressor Current Profiles (note cycling of #1 and #3)

Case Performance

Energy efficiency of cases is best achieved by the following:

- insuring proper temperature setpoints
- inspection of air curtain fan operation
- observation of proper load line limits
- maintaining minimal temperature gain to products throughout the cold train (i.e., the movement of product from delivery to storage to display)
- proper operation of case lighting and antisweat heaters.

Inspection of air curtain operation showed only three fans having problems - two were not functional at all and one was rather noisy due to probable imbalance or interference by a foreign object. These were reported to the contractor and were quickly repaired. DeCA stores experience a somewhat unique behavior with regard to load line limits - the extremely heavy traffic and short hours of operation require cases to be filled as high as possible prior to store opening (sometimes exceeding recommended load lines), and the cases are often close to empty before store closing. This presents an efficiency problem throughout the entire day, in that the air curtains are disrupted early when load lines are exceeded, throughout the day by heavy patron use (which is common), and later in the day due to the empty case. The latter condition is often not recognized as a problem, but air curtain performance is severely impaired by an empty case (in fact, manufacturers state that an empty case cannot be expected to achieve its design temperature. Air curtains are designed to be 'supported' by the products (or sometimes the shelving) within them. When a case is empty, the path of least resistance is for the air flow to fall into the empty case prior to reaching the suction grille at the bottom.

The walk-in freezer appeared to develop serious problems over the course of the summer. Moisture buildup in the freezer resulted in icy floors, hoarfrost collection on freezer walls and interior chainlink fence partitions, and total ice encasement of the unit fan coolers. Maintenance logs showed that the problem was originally diagnosed as a drain clogging problem during defrost. This was addressed, but the problem appeared to continue. This resulted in a major dispute which was ongoing between management and the contractor throughout the summer and into the fall. The situation was such that USACERL personnel felt that any involvement would be detrimental. The problem was ultimately solved, and was attributed to a fault in the unit cooler fan delay sequencing during defrost.

A number of case antisweat heaters appeared to be operating. They could most likely be disconnected due to the low ambient and in-store humidity at that location and due to the fact that a desiccant system is in operation and set to maintain the store relative humidity to 40% or lower. The former maintenance contractor could not be reached to check on why this situation existed and whether there was a problem with disconnecting antisweat devices. The current contractor was not familiar with whether the antisweat heaters were or were not connected.

Refrigerant Use

Refrigerant level checks were well documented by the previous contractor (Hussmann). Checks were made monthly from Dec 94 to May 96. Liquid receiver levels were used to indicate whether charge was required. A summary of refrigerant use through May 96 follows:

Rack 1 (R-502/404a;LT): No charge required between 12/94 and 5/96. Changed out to R-404a in July 96.
Rack 2 (R-502/404a;LT): 100 # R-22 added 3/26/96. No previous charge indicated.
Rack 3 (R-22; MT): 50# added 2/22/95
150# added 1/25/96 - leak repaired.
50# added 4/16/96 - no leaks indicated
Rack 4 (R22; MT): No charge required between 12/94 and 5/96.

Defrost Schedules and Performance

A defrost schedule is provided in Appendix A. A check of the control system indicated that the defrost schedule differed somewhat from that indicated. In addition, intermittent defrosts are often scheduled by the contractor as needed to alleviate frost problems due to case malfunction. The electric defrost system on medium temperature Rack 3 was monitored as a check on whether the proper number of defrosts are being scheduled. This can be determined in part by examining the length of time of typical defrosts versus the fail-safe time allotted. Table 4-1 provides a summary of the results, which are shown graphically in Figures C.7 - 17 through 19. There are 4 circuits which use electric defrost on Rack 3. Circuits 8 and 9 indicate operation as scheduled, at 1 and 4 times per day respectively. Circuit 10 and the beef staging unit coolers (12 & 13) have been altered from recommended practice. Circuit 10 has added one more defrost per day, while circuit 14 has doubled the number of defrosts from 2 to 4 per day. The high duty cycles indicates that this adjustment was warranted, but the reason for this should be investigated (particularly in a store with humidity control that rarely needs to operate). Local sources of humidity are probable causes - one should check for respirating loads in the circuit 10 case, and infiltration from the high humidity meat cutting area or an aggressive cleaning/hose-down schedule in the beef staging area could be the cause of the unit cooler defrost schedule.

Circuit	Load	Cycles/day	Time On	Fail Safe	Average	Duty
8	60' Meat (Top Display)	1	0000 hrs	36 min	21 min	58%
9	36' Dairy/Deli (Multideck)	4	0100 0700 1300 1900	36	32	89%
10	12' Meat/Deli (Multideck)	4* (3)	0200 0800 1400 2000	36	28	78%

14	Beef Staging (Unit Cooler 12-13)	4** (2)	0500 1100 1700 2300	36	34	94%

* Manufacturer and store design data indicate 3 defrosts per day recommended
** Manufacturer and store design data indicate 2 defrosts per day recommended

Table 4-1. Rack 3 Defrost Schedule and Performance

<u>Refrigeration System - Deficiencies and Recommendations:</u>

- Disconnect antisweat heaters or connect to RMCS to be controlled (if practical). This location should be able to operate without antisweat heaters.

- Program case lights to shut off during unoccupied hours (reactivate existing controls).

- Check on persistent problem reported in walk-in freezer. Investigate sources of moisture, and pressure differences between cooler and adjacent areas. Improper operation of exhaust fans may be a factor, especially EF-3 (1400 CFM) serving meat processing area.

- Set Danfoss control system to monitor compressor daily cycles. Check for leaking solenoids, missing or nonfunctional case fans on racks where compressors cycle consistently in excess of 60 times per day.

- Investigate circuits 10 and 14 for problems relating to excessive sources of moisture at or near the load.

- Investigate programmed defrost cycles and maintenance logs to determine if there is a problem with excess humidity at or inside the cases.

- Exhaust fan (EF-8) in mezzanine not operating - design intent calls for 85°F maximum space temperature setpoint. Excessive room temperatures can decrease system efficiency due to inadvertent heating of refrigerant stored in the liquid receiver.

5.0 Summary of Major Deficiencies Identified

Numerous small deficiencies were identified and either corrected on-site or reported to the contractor and fixed through the maintenance program. Examples of small deficiencies and adjustments not detailed here include worn walk-in door seals, 2 case fan malfunctions, replacement of well thermometers, reopening and verifying performance of the hot water coil to the main DD-1 air handler, and adjustment of valve actuator and control throttling range (gain) on hot water coil. A rather large problem with the packaged chiller was reported to Scott-Polar during instrumentation phase, with data contributed to help diagnose an immediate problem (extremely excessive compressor cycling).

The following summarizes the most substantive deficiencies that were not corrected by USACERL during the investigation.

Summary: Action Items

1. Negotiate lower flat-rate electricity charge with Academy Energy Office.
2. Reduce DD-1 supply air CFM.
3. Implement DD-1 night setback.
4. Restore heat reclaim function to DD-1 coils.
5. Investigate economizer operation of DD-1 (never implemented).
6. Repair DD-1 fire damper actuator and linkages and PVT fire suppression system.
7. Increase deadband on heating and cooling of DD-1 zone.
8. Repair desiccant unit:
    a. Repair tubing from flow proof switch 1AFS.
    b. Correct fan sequence logic on desiccant unit (if needed).
9. Recalibrate humidistat and lower RH setpoint to 35%.
10. Implement night/off-hour setback on admin zone AH-2.
11. Implement economizer control on AH-2.
12. Consider installing dedicated rooftop DX unit for AH-2.
13. Get chilled water under on/off control by implementing OA lockout. This strategy will substantially reduce compressor cycling. Also, night setback of AH-1 and AH-2 will contribute to reducing cycling.
14. Investigate additional strategies for optimizing oversized chiller operation.
15. Address oversizing of packaged chiller. Consult factory rep for potential short-term operational scenarios prior to startup next season.
16. Downsize chilled water pumps (or trim impeller). Dependent on package chiller oversizing decision.
17. Get boilers under on/off control by implementing OA lockout.
18. Implement lead/lag strategy for boilers.
19. Implement hot water reset strategy to save heating energy during spring and summer operation.
20. Implement boiler staging (upon boiler replacement).
21. Trim impellers on HWS to match maximum load requirement.

22. Repair condensate leak at boiler B-2 exhaust.
23. Disconnect antisweat heaters (contractor had conflicting information here).
24. Shut off case lights during unoccupied hours (reactivate existing controls).
25. Set Danfoss control system to monitor compressor daily cycles.
26. Exhaust fan (EF-8) in mezzanine not operating. Relay disabled. Investigate reason and reconnect.
27. Investigate circuits 10 and 14 for problems relating to excessive sources of moisture at or near the load.
28. Reduce light levels during off hours.

Summary by Deficiency Type (no particular order)

Design Deficiencies

- Chiller design oversized (30 ton required; 50 ton designed; 65 ton installed).
- Small admin and bakery zones should be on separate AC unit.
- Install economizer operation to admin and bakery zone air handlers.
- Desiccant system economics does not warrant initial cost.
- Boilers should be specified to allow for part load operation.
- Hot water system OA reset should be employed.
- Main store supply air CFM oversized.
- Continuous supply fan operation in admin zone unnecessary.

Construction Deficiencies

- Installed chiller increased further from oversized 50 ton to 65 ton.
- Downsize motor/pump or trim impellers on CHWS pumps.
- Trim impellers on HWS pumps.

Operational Deficiencies

- Night setback not functional - main store (DD-1).
- Night setback not functional - administrative zone (AH-2).
- Heat reclaim not functional.
- Reduce store light levels during off hours.
- Recalibrate humidistat and lower RH setpoint to 35%.
- Increase deadband on heating and cooling of DD-1 zone.
- Relay disabled on exhaust fan (EF-8) in mezzanine.
- Set Danfoss control system to monitor compressor daily cycles.

- Shut off case lights during unoccupied hours (reactivate existing controls).
- Disconnect antisweat heaters (contractor had conflicting information here).
- Implement lead/lag strategy for boilers.

Maintenance Deficiencies

- Check for cracked pneumatic line on proof switch 1AFS (desiccant system)
- Correct fan sequence logic on desiccant unit (if above not source of desiccant problem).
- Repair condensate leak at boiler B-2 exhaust.
- Repair DD-1 fire damper actuator and linkages and PVT fire suppression system.

6.0 Action Plan for Addressing Recommissioning and Implementation of Other ECOs

An economic analysis of the impacts of many of the above repairs and improvements would require the use of simulation tools. Even then, simulations are known to fall short when trying to model actual mechanical system behavior or assess the impacts of various systems changes and improvements.

Most changes suggested by this report are extremely cost-effective, or are warranted as comfort, productivity, or operation and maintenance projects. The costs of implementation, however, should not be considered 'free'. One factor affecting cost is dependent on the maintenance contract requirements. Another is the degree of field work needed to make the recommended change, and then to assess the performance of the change afterward to confirm proper operation. Finally, there is a feedback to the contractor to confirm that the changes are understood and can be maintained.

The following parties need to jointly determine the appropriate scope for executing the recommissioning effort:

- DeCA HQ
- store personnel
- contract maintenance
- commissioning agent (in this case USACERL)

Disagreements between parties regarding intended operation or ability of systems to sustain operation as designed often results in the building reverting back to it's inefficient or dysfunctional state. The extent to which the various parties should agree should be determined by DeCA HQ. Experience with successful recommissioning (and subsequent implementation of ECO projects) dictates that the following items should be required:

1. Working description and schematics of all equipment and systems in place.
2. Working description of proper sequence of operation after ReCX.
3. Complete all deficiency action items agreed upon in the scope. This MUST be done to the extent practical by the commissioning agent and contract maintenance group, possibly with support from manufacturer's personnel.
4. Verification of working systems (including data collection or trend analysis plots).
5. Supply any training needed or requested by any parties involved in keeping the store functioning efficiently.

Items 1 and 2 become a working document which the contractor can consult often as needed. THIS DOCUMENT MUST INCLUDE CONTRACTOR INPUT. It should be available in the equipment room at all times. Phone numbers of major equipment vendors should be included, including the extension of service technicians and applications engineers. The use of free advice from manufacturers field support is underused, particularly with regard to HVAC and controls equipment.

The role of the contract maintenance provider is crucial to efficient commissary operations. Their job can be very difficult, especially given the number of different types of stores, associated equipment and interacting systems they are tasked with keeping in operation. Their role - by definition or by necessity - becomes one of keeping the cases at proper temperature and the thermal conditions acceptable to the occupants. Discussions with various DeCA contractors indicates that they are often confused regarding the specific equipment, intended operation, and history of problems when called in on an emergency. Some contractors use 'first available' personnel, which can result in different workers working on systems within the same store. The magnitude of this problem will vary between contractors and even from location to location. Use of equipment room maintenance logs help alleviate these problems to the extent that they are used. Entries must be made regularly and consistently, and a review of logged entries should be made on a regular basis. The use of an automated contract maintenance system may be warranted. These systems have been shown to dramatically improve the ability of maintenance managers and field personnel to diagnose and solve problems.

Finally, the need for feedback of system operation to maintenance personnel is more critical than ever for today's advanced energy efficient systems. Ideally, the RMCS system would be designed, specified, and operated to easily allow simple plots of performance parameters to be pulled down. Many performance problems obtained during the course of this study are not obvious to maintenance workers visiting the store on 'equipment fault' calls. Preventative maintenance (PM) could allow the contractor the time to investigate energy efficient performance, but is more typically geared toward refrigerant and lubricant checks, cleaning tasks. One PM checklist reviewed by USACERL (supplied by Nelson Refrigeration, Omaha, NE) showed a rather complete inventory of items, but many would require more time to complete than is typically available or would be difficult to answer correctly without the necessary historical data.

Appendices

Measured Systems Performance and Diagnostic Testing Report

Commissary
USAF Academy, Colorado

Appendix A: Equipment Schedules Obtained on Site

Appendix B: Datalogger Plan and Schedule

Appendix C: Measured Data

Appendix D: Equipment Summary

Prepared by:

U.S. Army Construction Engineering Research Laboratories
Champaign, IL

Appendix A

Equipment Schedules Obtained on Site

1) HVAC Sequence of Operation (Design)

2) Refrigeration Design Summary
   - Rack Descriptions
   - Case and Circuit Schedule
   - Defrost Schedule

Appendix B

Datalogger Plan and Schedule

Appendix C

Measured Data (grouped by system)

1) Weather Data

2) Main Store Zone Air Handling Unit w/Desiccant (DD-1)

3) Main Store Zone - Space Conditions

4) Administrative Zone Air Handling Unit (AH-2)

5) Chilled Water Supply System
    - Packaged Chiller
    - Circulation Pumps

6) Hot Water Supply System
    - Boiler 1
    - Boiler 2
    - Circulation Pumps and Valves

7) Commercial Refrigeration System
    - Rack 1 (Low temp)
    - Rack 2 (Low temp)
    - Rack 3 (Medium temp)
    - Rack 4 (Medium temp)
    - Defrost

Appendix C.1 - Weather Data

Appendix C.2 - Main Store Zone Air Handling Unit w/Desiccant (DD-1)

Appendix C.3 - Main Store Zone - Space Conditions

Appendix C.4 - Administrative Zone Air Handling Unit (AH-2)

Appendix C.5 - Chilled Water Supply System

- Packaged Chiller
- Circulation Pumps and Valves

Appendix C.6 - Hot Water Supply System

- Boiler 1
- Boiler 2
- Circulation Pumps and Valves

Appendix C.7 - Commercial Refrigeration System

- Rack 1 (Low temp)
- Rack 2 (Low temp)
- Rack 3 (Medium temp)
- Rack 4 (Medium temp)
- Defrost

Appendix D

Equipment Summary

Boilers (2)

Type: Low Pressure (Packaged)
Manufacturer: Lochinvar Corp.,
1930 Air Lane Dr.,
Nashville, TN 37210
(800)-722-2101
Model: PBN1000 A
Input: 1,000,000 Btu
(800,000 Btu/h derated)
Output: 880,000 Btu
(704,000 Btu/h derated)

CHW Pump/Motor Assembly (2)

Pump: Aurora/General Signal
Model: Centrifugal 91-154-28-1
Type: 341A-DF
GPM: 115
Head: 93 ft
RPM: 1750
Motor: GE 5K213JD221A
HP: 7.5
Phase: 3
Amp: 22.8/11.6
Volts: 230/460
Efficiency: 85.5% (NEMA nominal)
SF: 1.15

HWS Pump/Motor Assembly (2)

Pump: Aurora/General Signal
Model: Centrifugal 91-154-02-1
Type: 341A-DF
GPM: 167
Head: 99 ft
RPM: 1750
Motor: GE 5K213JD221A
HP: 7.5

Phase: 3
Amp: 22.8/11.6
Volts: 230/460
Efficiency: 85.5% (NEMA nominal)
SF: 1.15

<u>Packaged Chiller</u>
Mfr.: Dunham-Bush
Model: NC65AQ
Serial No.: 71705001A92C
Overall Capacity: 65 ton
No. Compressors: 2 (30 hp, 35 hp)
Refrigerant: R-22
Sequence: 0, 23, 46, 77, 100%

Compressors

ID: CH-1 (Lead compressor)
Mfr.: Copeland
Model: Copelametic Discus 4DR3-3000-TSK
Serial: CT_92F_02491_S
Volts: 208-230/460
RLA: 94/47
Capacity: 30 hp
Measured Power: 23.4 kW (31.4 hp)
Date: 1992
Note: This compressor was replaced in first year of commissary operation. Has cylinder unloader.

ID: CH-2 (Lag compressor)
Mfr.: Copeland
Model: Copelametic 6DH1-3500-TXK
Serial: CT_92A_04602_S
Volts: 208-230/460
RLA:
Capacity: 35 hp
Measured Power: 27.4 kW (36.8 hp)
Date: 1992

Condenser:

Model: AD 80
Serial: 71705002A92C
Fans: 8

<u>DHW Boiler</u>

Mfr.: Raypack

DD-1 Air Handling Unit (Main Store Zone)
Manufacturer: Munters
Model: SuperAire S-30

Main Supply Fan/Motor:
Mfr.: A. O. Smith
Model: OT04004
Motor Capacity: 40 HP
Motor RPM: 1725
Fan RPM: 1357
Fan Size: 330
Fan Type: BAF Class I
Nom. Eff.: N/A
Nameplate Amps: 10.7-10.0/5.0
Volts: 208-230/460
Phase: 3
Volume: 26,000 CFM design
Static pressure: 3.30” w.c.

Desiccant Reactivation Fan/Motor:
Mfr.:
Model:
Motor Capacity: 7.5 HP
Motor Type: TEFC
Motor RPM: 3450
Amps: 13.4/6.7
Volts: 230/460
Phase: 3
Nom. Eff.: 82.5%
Volume: 3705 CFM
Fan Size: M 15
Fan Type: BAF Class I
Static pressure: 3.15” w.c.

Desiccant Process Fan/Motor:
Mfr.: World Energy
Model: OT04004
Motor Capacity: 5 HP
Motor Type: ODP
Motor RPM: 3460
Nom. Eff.: 82.5%
Fan Manufacturer: Delhi

Fan Size: 918
Fan RPM: 760
Fan Type: FC Class I
Volts: 230/460
Amps: 19.4/9.7
Phase: 3
Volume: 6,890 CFM
Static pressure: 1.92” w.c.

Reactivation Duct Furnace
Manufacturer: Sterling
Fuel: NG
Capacity: 400,000 Btu/h (input)
320,000 (input derated)
308,000 Btu/h (output)
246,400 (output derated)

DD-1 Chilled Water Coil
Manufacturer.: Heatcraft
Location: DD-1 supply duct
Type: Finned tube (al/cu; 6 fpi)
Overall Size: 72”h x 77” x 4 rows
Capacity: N/A

DD-1 Hot Water Coil
Manufacturer.: Heatcraft
Location: DD-1 supply duct
Type: Finned tube (al/cu; 14 fpi)
Overall Size: 72”h x 77” x 1 row
Capacity: N/A

DD-1 Heat Reclaim Condensers
Manufacturer.: Heatcraft
Location: Reactivation
Type: Finned tube (al/cu; 14 fpi)
Overall Size: 24”h x 40” x 8 rows
Capacity: 100,000 Btu/h
Location: DD-1 supply duct
Type: Finned tube (al/cu; 9 fpi)
Overall Size: 72”h x 77” x 4 rows
Capacity (est.): 300,000 Btu/h

<u>Administrative Area Air Handling Unit (AH-2)</u>
Manufacturer: Dunham-Bush
Model: HCSO3LF71705202

Serial: 717052-02A92C
Rating: 1570 CFM @ 1.97” t.s.p.
Motor: Baldor
  RPM: 1676
  Type: ODP
  Capacity: 1 hp
  Rated Voltage: 208-230/<u>460</u>
  Rated Amps: 3.6-3.4/<u>1.7</u>
  Measured power: 1.28 kW
Actual CFM: N/A

<u>LT Rack 1A & B</u>
Mfr.: Tyler
Model: Equalizer
Compressor Mfr.: Carlyle
No. Compressors: 4
Compressor Capacities:
  1-7.5 hp (LTPCR-751-502L)
  1-10 hp (LTPCR-1001-502L)
  1-20 hp (LTPCR-2001-502L)
  1-10 hp (THBCR-1001-502L; Booster)
No. Circuits: 4
Circuit Design Load: 160,400 Btu/h
Circuit Design Temperature: -25°F
(Booster to -35°F)

<u>LT Rack 2A & B</u>
Mfr.: Tyler
Model: Equalizer
Compressor Mfr.: Carlyle
No. Compressors: 3
Compressor Capacities:
  1-10 hp (LTPCR-1001-502L)
  1-20 hp (LTPCR-2001-502L)
  1-10 hp (THBCR 1001-502L; Booster)
No. Circuits: 4
Circuit Design Load: 126,100 Btu/h
Circuit Design Temperature: -25°F

<u>MT Rack 3</u>
Mfr.: Tyler
Model: Equalizer
Compressor Mfr.: Carlyle

No. Compressors: 4
Compressor Capacities:

- 1-5 hp (MTPCR-501-22S)
- 1-7.5 hp (MTPCR-761-22S)
- 1-10 hp (MTPCR 1001-22S)
- 1-15 hp (MTPCR 1501-22S)

No. Circuits: 10
Circuit Design Load: 334,400 Btu/h
Circuit Design Temperature: +15°F

MT Rack 4
Mfr.: Tyler
Model: Equalizer
Compressor Mfr.: Carlyle
No. Compressors: 4
Compressor Capacities:

- 1-5 hp (MTPCR-501-22S)
- 1-7.5 hp (MTPCR-761-22S)
- 1-10 hp (MTPCR 1001-22S)
- 1-25 hp (MTPCR 2501-22S)

No. Circuits: 10
Circuit Design Load: 450,500 Btu/h
Circuit Design Temperature: +20°F